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INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(51) International Patent Classification 7 : G05D 7/01

A1

(11) International Publication Number: WO 00/26736

(43) International Publication Date: 11 May 2000 (11.05.00)

(21) International Application Number: PCT/GB99/03597

(22) International Filing Date: 3 November 1999 (03.11.99)

(30) Priority Data:
9823927.0 3 November 1998 (03.11.98) GB

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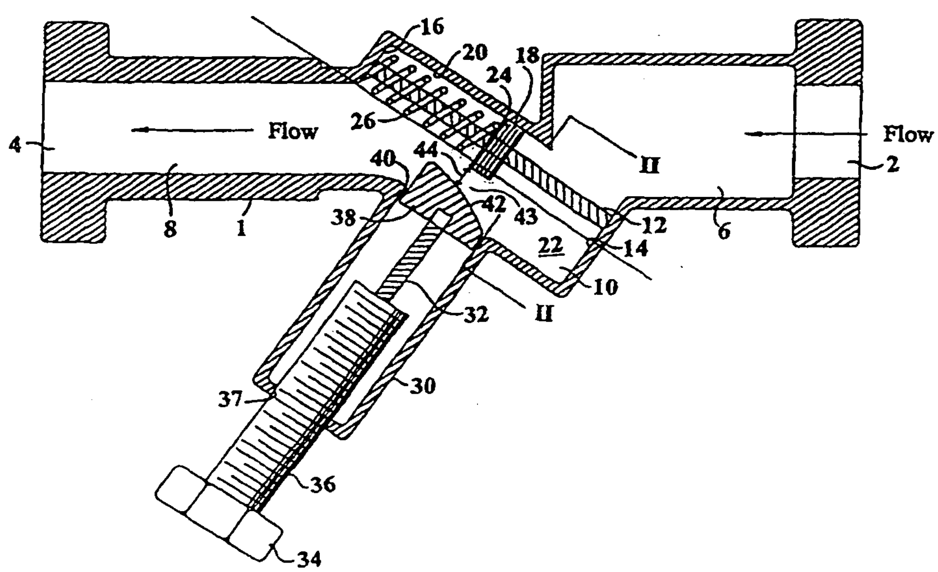
(81) Designated States: AE, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, CA, CH, CN, CR, CU, CZ, DE, DK, DM, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, NO, NZ, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, US, UZ, VN, YU, ZA, ZW, ARIPO patent (GH, GM, KE, LS, MW, SD, SL, SZ, TZ, UG, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GW, ML, MR, NE, SN, TD, TG).

Published
With international search report.

(54) Title: CONSTANT FLOW VALVE



(57) Abstract

A constant flow valve includes a housing (1) containing a fluid flow passageway (10), a resiliently-biased piston (18) located in said passageway and mounted for movement in response to changes in the differential fluid pressure across the valve, and a throttle member (38) located in the passageway in the vicinity of the piston to define a throttle passageway between the piston and the throttle member. The throttle passageway (43) is tapered and defines at its point of minimum size a throttle orifice (44). The size of said throttle orifice varies in use with movement of the piston (18) in response to changes in the differential pressure, such that the flow rate of fluid through the valve is substantially constant.

CONSTANT FLOW VALVE

The present invention relates to a constant flow valve for delivering a substantially constant flow rate of fluid irrespective of the differential pressure across the valve. In particular, but not exclusively, the invention relates to an externally adjustable constant flow valve, whereby the flow rate through the valve may be adjusted.

For convenience, the invention will be described with particular reference to an application in a re-circulating hot water radiator central heating system, but it will be appreciated that its uses are not so limited and, indeed, it has wide applicability in fluid systems generally.

It is conventional practice to use calibration valves to balance the distribution of flows in large central heating systems such as in a multi-storey office block. A primary pipe loop re-circulates pumped heated water from the boiler usually located in the basement to the uppermost floor. At each floor a secondary piping loop is directly connected to the primary loop and feeds a series of radiators connected between the supply and return pipes of the secondary loop. Tertiary pipe loops may also be connected to secondary loops, and so forth. Clearly, the differential pressure across any pipe loop is dependent upon the height of the entry and exit points from the boiler and the individual pipe run friction losses. In addition to the difficulties caused by this, the flows required to meet the heating requirements on each floor are not necessarily the same and indeed may even change on a daily basis as a function of individual requirements (turning radiators on and off) or occupancy.

To obtain the desired distribution of flows necessitates the use of balancing valves, which are usually fitted between the last radiator and the connection from the return pipe of the secondary loop to the primary loop, or the tertiary pipe loop to the secondary and so forth. To set the design flow manually the degree of throttling of any particular balancing valve necessitates that a flow meter is also installed in the pipe loop. The flow meter is usually but not exclusively an integral part of the balancing valve connected by way of differential pressure tappings across the balancing valve. However, adjusting the setting of any one valve affects the differential pressure across all the other valves, thus manual adjustment is both time-consuming and inaccurate. Furthermore, if a change occurs in either the head

flow characteristic of the pump or the individual friction resistance of any of the individual pipes, this too may alter the optimal setting of one or all of the balance valves.

An alternative and preferable approach is to use constant flow valves. A common arrangement uses a variable orifice set against a spring so that the differential pressure determines the degree of occlusion across the variable orifice. A more complex arrangement takes the form of an automatic self-energising control valve. In one such variable orifice type valve, the variable orifice is formed in the side wall of a spring-biased piston, which moves relative to a sleeve according to the differential pressure. The orifice area is often divided into a front facing fixed orifice and one or more side orifices such that the combined variable discharge area yields the design flow over the required range of differential pressures. This yields both primary and secondary flow paths. When the differential pressure is low, a large discharge area is provided and when the differential pressure is high, the spring is compressed and the sleeve partially occludes the orifice, thereby maintaining a substantially constant flow rate. The piston and the spring may be provided in the form of a cartridge that can be removed from the valve body and replaced with another cartridge providing a different flow rate.

A number of problems exist with this arrangement: first, for low and very low flows the Reynolds numbers are in the lamina or transitional regime of flows, which can cause a lack of repeatability due to the variability in the profile of the approach flow. Second, the variable occlusions machined in the side walls to provide the required constant flow rates necessitate very accurate machining. In conventional form this approach also necessitates that an individual and precise geometry of the variable occlusions are required for any given flow. due to the existence of one or more flow paths through the piston occlusions the division of flow between the paths is not necessarily repeatable and therefore this arrangement tends to lead to hysteresis between rising and falling secondary pipe resistances. The use of fixed geometrical shapes invariably leads to an engineering compromise in the range of constant flows that can be economically afforded in any one size of valve. Further, the flow rate cannot be adjusted externally, it being necessary to replace the cartridge in order to provide a different flow rate. A further disadvantage exists in that it becomes necessary to remove the cartridges containing the precise geometrical shapes on

commissioning to avoid the problem of debris being trapped in the variable orifices and therefore rendering the balancing valves ineffective.

It is an object of the present invention to mitigate at least some of the aforesaid disadvantages.

According to the present invention there is provided a constant flow valve including a housing containing a fluid flow passageway, a resiliently-biased piston located in said passageway and mounted for movement in response to changes in the differential fluid pressure across the valve, a throttle member located in the passageway in the vicinity of the piston to define a throttle passageway between the piston and the throttle member, said throttle passageway being tapered and defining at its point of minimum size a throttle orifice, wherein the size of said throttle orifice varies in use with movement of the piston in response to changes in the differential pressure, such that the flow rate of fluid through the valve is substantially constant.

The flow rate of fluid through the valve is of course substantially constant only for variations in the differential pressure that lie within a predetermined range: i.e. between upper and lower operational limits, for example from 10kPa to 250kPa, or from 30kPa to 450kPa, depending on the chosen design characteristics of the valve. The statement that the flow rate is "substantially constant" implies that the flow rate is regulated to within a tolerance of, for example, ±5%.

The valve does not rely upon the use of one or more precisely machined geometrically complex shaped side orifices and can therefore be manufactured more cheaply than existing constant flow valves. Also, as the valve only requires a single orifice, the hysteresis effects caused by cascading flows are avoided.

Advantageously, the taper of the throttle passageway is non-linear. For example, the profile of the passageway may be shaped to match the relationship between ceiling height and relative piston displacement for a constant flow rate, as depicted in Figure 3 of this application.

The constant flow valve may include a flow adjusting means for adjusting the flow rate of fluid through the valve. The provision of means for adjusting the flow rate through the

valve allows the flow rates through different parts of a complicated fluid flow system, such as a central heating system, to be balanced relatively easily. Preferably, the flow adjusting means is arranged to allow for infinitesimal adjustment of the flow rate.

Advantageously, the flow adjusting means is connected to the throttle member and is operable to adjust the position of the throttle member relative to the piston. Preferably, the throttle member is movable in a direction substantially perpendicular to the direction of movement of the piston. This provides a relatively simple and reliable mechanism for adjusting the flow rate.

Alternatively, the flow adjusting means may be connected to the piston and operable to adjust the position of the piston relative to the throttle member. The piston may be adjustable in a direction substantially parallel to the direction of movement of the piston.

The constant flow valve may include a profile adjusting means for adjusting the profile of the tapered throttle passageway, to provide a substantially constant flow over a wide range of different flow rates and differential pressures. Advantageously, the profile of the tapered throttle passageway is adjusted by adjusting the angle of the throttle member. Preferably, the profile adjusting means is arranged to adjust the angle of the throttle member as the position of the throttle member is adjusted, so that profile of the throttle passageway is automatically adjusted for the preset flow rate.

Alternatively, the profile of the tapered throttle passageway may be adjusted by adjusting the angle of the piston, and the profile adjusting means may be arranged to adjust the angle of the piston as the position of the piston is adjusted.

Embodiments of the present invention will now be described, by way of example, with reference to the accompanying drawings, in which:

Figure 1 is a side section of a first type of valve according to the invention;

Figure 2 is a cross-section on line II-II of the valve shown in Figure 1;

Figure 3 is a graph illustrating the ceiling profiles required to provide a constant flow at various different flow rates;

Figure 4 is a side section of a second type of valve according to the invention;

Figure 5 is a cross-section on line V-V of the valve shown in Figure 4;

Figure 6 is a side section of a third type of valve according to the invention;

Figure 7 is a cross-section on line VII-VII of the valve shown in Figure 6;

Figure 8 is a side section of a third type of valve according to the invention;

Figure 9 is a cross-section on line IX-IX of the valve shown in Figure 8;

Figure 10 is a cross-section showing a modified form of the valve shown in Figure 8; and

Figure 11 is a side section of a fourth type of valve according to the invention.

The first type of valve, which is shown in Figures 1 and 2, includes a housing 1 having an inlet port 2 and outlet port 4. A flow passageway extends from the inlet port to the outlet port, the passageway including an inlet portion 6 and an outlet portion 8, both of which are circular in cross-section, and an intermediate portion 10 that is rectangular in cross-section. The central axis of the intermediate portion 10 is placed at an acute angle to the central axes of the inlet and outlet portions 6,8 and the upstream end of the intermediate portion 6 is displaced laterally from the central axis of the inlet 6 so that water flowing through the valve experiences two abrupt changes in direction before entering the intermediate portion 10.

A circular shaft 12 is mounted in the intermediate flow passage portion 10 between the first and second end walls 14,16 of that portion, and extends parallel to the longitudinal axis of the intermediate portion.

A piston 18 is mounted on the shaft 12 for sliding movement along the shaft. The piston 18 has a rectangular cross-section and has a close fit with the lower wall 20 and the lower portions of the two side walls 22 of the intermediate portion 10 (as shown most clearly in Figure 2). The two side walls and the bottom wall of the piston 18 are provided with a series of parallel grooves 24 that extend around those walls. In use, these grooves lead to the formation of trapped eddies and reduce leakage along the side walls.

A compression spring 26 is mounted on the shaft 12, the ends of the spring bearing against the second end wall 16 and the underneath of the piston 18, thereby urging the piston 18 towards the first end wall 14 against the flow of water through the valve. In use, the position of the piston depends on the differential pressure across the valve.

The housing 1 includes an arm 30 having a square cross-section that extends perpendicularly from the intermediate portion 10 of the flow passage. A rod 32 extends along the longitudinal axis of the arm. The outer end of the rod 32 is engaged by an adjusting screw 34 having an external screw thread 36 that engages a threaded bore 37 in the end wall of the arm 30. Rotating the adjusting screw 34 drives the rod 32 backwards or forwards along the axis of the arm.

The inner end of the rod 32 is attached to a throttle plate 38 that has a square cross-section and is mounted for sliding movement along the arm 30. A sealing member 40, for example an O-ring, extends around the four sides of the plate to seal the gap between the plate and the four walls of the arm 30.

The face 42 of the plate 40 that is nearest to the piston 18 has a curved convex shape and defines with the piston 18 a tapered throttle passageway 43. The throttle passageway 43 tapers in the downstream direction and is narrowest at the rear edge of the piston 18. The gap between the rear edge of the piston and the curved face of the throttle plate 38 forms a throttling orifice 44 that, in use, controls the flow rate of fluid through the valve.

In operation, the position of the piston 18 is determined by the differential pressure across the valve. When the pressure is low, the piston 18 will be positioned close to the first end wall 14 of the intermediate flow passage portion 10 and the throttling orifice 44 between the piston and the throttle plate 42 will therefore be large. At high differential pressures, the piston 18 will be pushed towards the second end wall 16 of the intermediate flow passage portion 10, compressing the spring 26. The throttling orifice 44 between the piston 18 and the throttle plate 42 will then be small. Because the orifice is large at low differential pressures and small at high differential pressures, the flow rate of fluid through the valve will be substantially constant, regardless of the differential pressure across the valve (providing that the differential pressure is within a predetermined operational range).

In order to reduce the flow rate, the adjusting screw 34 is turned, moving the throttle plate 42 closer to the piston 18. This reduces the size of the throttling orifice 44 thereby reducing the flow rate through the valve. The use of an adjusting screw allows for infinitesimal adjustment of the flow rate.

To increase the flow rate, the adjusting screw 34 is turned the opposite way, moving the throttle plate 38 further from the piston 18 and increasing the size of the throttling orifice 44. The flow rate through the valve is thereby increased.

Upon commissioning, the valve may be fully opened to avoid the risk of the valve becoming blocked by debris within the piping system.

The operation of the valve is illustrated graphically in Figure 3. In this graph, the vertical axis represents the distance between the rear edge of the piston 18 and throttle plate 38 (the "ceiling height"), which is proportional to the area of the throttling orifice 44. The horizontal axis represents the displacement of the piston from the rest position (the position when the differential pressure across the valve is zero), which is proportional to the differential pressure. The curved lines represent the relationship between these two quantities for different flow rates. These curves therefore represent the shape that the curved face 42 of the throttle plate 38 must have to produce different constant flow rates.

It will be seen that for large piston displacements (corresponding to high differential pressures) the curves are similar in shape. This suggests that for this range of differential pressures, a substantially uniform flow rate will be provided at a wide range of flow rate settings. However, at lower displacements (corresponding to low differential pressures) the curves tend to diverge, suggesting that a profile designed to deliver a particular flow rate might not have the perfect shape for higher or lower flow rates, particularly at low differential pressures. Although this is not ideal, for many situations the flow rate may be sufficiently uniform for practical purposes.

A second type of constant flow valve is shown in Figures 4 and 5. This is similar in many respects to the first valve shown in Figures 1 and 2 and where the components of the two valves are similar, the same reference numerals have been used. The most significant differences lie in the shapes of the piston 50, which is substantially semi-cylindrical, and the

lower wall 52 of the intermediate flow passage portion 10, which is also semi-cylindrical to match the shape of the piston 50. The piston 50 is mounted for sliding movement on the shaft 12 and is provided with a key-way 54 in its cylindrical surface that engages a key 56 to prevent rotation of the piston 50. Grooves 58 extend around the cylindrical face of the piston 50, which trap eddies to reduce leakage past the sides of the piston 50.

A compression spring 60 is located between the underside of the piston 50 and the second end wall 16 of the intermediate flow passage portion 10, which urges the piston 50 towards the first end 14 of the intermediate flow passage portion, against the flow of fluid through the valve.

The valve operates in a similar manner to the first valve shown in Figures 1 and 2: when the differential pressure across the valve increases, the flow of fluid pushes the piston 50 towards the second end wall 16, compressing the spring 60. This reduces the size of the orifice 44 between the piston 50 and the throttle plate 38, so preventing any significant increase in the flow rate of fluid through the valve. When the differential pressure decreases, the spring 60 urges the piston towards the first end wall 14, increasing the size of the orifice 44 and again, therefore, maintaining a constant fluid flow rate.

A third type of constant flow valve is shown in Figures 6 and 7. This is similar is most respects to the second valve shown in Figures 4 and 5 and where the components of the valve are similar to those of the second valve, the same reference numerals have been used.

The valve differs from the second valve in that the throttle plate 70 is attached to the rod 32 by means of a pivot joint 72 that allows the throttle plate 70 to rotate about an axis that is perpendicular to the longitudinal axes of the rod 32 and the shaft 12. A control pin 74 is connected at one end to a wall of the arm 30 by means of a pivot joint 76, and engages an angled slot 78 provided in the rear face of the throttle plate 70 to control the pivoting movement of the throttle plate 70,. A sealing flange 80 prevents fluid flowing into the arm 30 past the sides of the throttle plate 70.

When the adjusting screw 34 is turned to move the throttle plate 70 towards the piston 50, the throttle plate rotates about the pivot joint 72 under the control of the control pin 74, reducing the taper of the throttle passageway 43 between the throttle plate 70 and the piston

50. When the adjusting screw 34 is rotated in the opposite direction to move the throttle plate 70 away from the piston 50, the throttle plate 70 pivots the other way, increasing the taper of the throttle passageway 43. In this way, the effective contour of the throttle plate may be adjusted to provide a substantially constant flow rate at all flow rate settings.

A fourth type of constant flow valve is shown in Figures 8 and 9. This is similar in many respects to the second valve shown in Figures 4 and 5 and where the components of the valve are similar to those of the second valve, the same reference numerals have been used. In particular, the valve includes an adjusting screw 34 that is connected via a rod 36 to a throttle plate 38 that mounted for sliding movement towards and away from a piston 90. The throttle plate 38 has a curved convex shape and defines with the piston 90 a tapered throttle passageway 43 and a throttling orifice 44.

The most significant difference lies in the fact that the position of the piston 90 may be adjusted longitudinally. The piston 90 is mounted for sliding movement along a shaft 92 and is provided with a key-way 94 that engages a key 96 to prevent rotation of the piston 90. Grooves 98 extend around the cylindrical face of the piston 90, which trap eddies to reduce leakage past the sides of the piston 90.

A compression spring 100 is located between the underside of the piston 90 and a plate 102 that is mounted on the shaft 92 adjacent the second end wall 16. The plate engages a screw thread 104 provided on the shaft 92 and is provided with a key-way that engages the key 96 to prevent rotation of the plate. The shaft 92 extends through the first end wall 14 and an adjusting knob 106 is provided on the projecting end of the shaft. Using this knob, the shaft 92 may be rotated, causing the plate 102, the spring 100 and piston 90 to move longitudinally along the shaft 92.

It will be observed that the upstream end 108 of the key 96 is curved (the curvature being exaggerated in the drawings) and that the piston 90 is mounted on the shaft 92 by means of a ball joint 110. This causes the piston to rotate through a few degrees when it is positioned towards the upstream end of the shaft.

The arrangement shown in Figs. 8 and 9 allows the relative positions of the piston and the throttle plate 38 to be adjusted both the longitudinally and the transversely, while

simultaneously adjusting the taper of the throttle passageway 43. By adjusting these three quantities, it is possible to increase the accuracy of the valve, allowing it to provide a substantially uniform flow over a wide range of flow rates.

The arrangement shown in Fig, 10 is similar to that shown in Figs. 8 and 9, except that the piston 120 has a rectangular cross-section.

Fig. 11 shows a further arrangement in which the valve has an in-line construction. As with the valve shown in Figs. 8 and 9, the valve includes an adjusting screw 134 that is connected via a rod 136 to a throttle plate 138 that is mounted for sliding movement towards and away from a piston 140. The throttle plate 138 has a curved convex shape and defines with the piston 140 a tapered throttle passageway 142 and a throttling orifice 144.

The position of the piston 140 may be adjusted longitudinally. The piston is mounted for sliding movement along a shaft 146 and is provided with a key-way 148 that engages a key 150 to prevent rotation of the piston 140. A compression spring 152 is located between the underside of the piston 140 and a plate 154 that is mounted on the shaft 146 adjacent the second end wall 156. The plate engages a screw thread 158 provided on the shaft 146 and is provided with a key-way that engages the key 150 to prevent rotation of the plate. The shaft 146 extends through the first end wall 160 and is provided with an external gear 162 that is engaged by a rotatable crown wheel 164, mounted around the inlet end of the valve housing 166. Using this crown wheel, the shaft 146 may be rotated, causing the plate 154, the spring 152 and piston 140 to move longitudinally along the shaft 146.

The upstream end 168 of the key 150 is curved (the curvature being exaggerated in the drawings) and the piston 140 is mounted on the shaft 146 by means of a ball joint 170. This causes the piston to rotate through a few degrees when it is positioned towards the upstream end of the shaft.

The arrangement shown in Fig. 11 allows the relative positions of the piston 140 and the throttle plate 138 to be adjusted both longitudinally and transversely, while simultaneously adjusting the taper of the throttle passageway 142. By adjusting these three quantities, it is possible to increase the accuracy of the valve, allowing it to provide a substantially uniform flow over a wide range of flow rates.

The valves shown in Figs. 8-11 may be modified to include a link (for example a mechanical link) between the two adjusting knobs, so that the longitudinal and transverse positions of the piston and the throttle plate may be adjusted simultaneously.

Various modifications of the valve are possible, some of which are described below.

The spring 26,60 may be replaced by another form of resilient biassing means, for example using an elastic material or a cylinder of compressed gas. Also, the resilient biassing means may be arranged to provide non-linear biasing characteristics: for example, a variable rate spring may be used.

Instead of using a control pin 74, the angle of the throttle plate 70 of the valve shown in Figures 6 and 7 may be controlled by providing lugs on either side of the throttle plate, which engage contoured grooves in the side walls of the arm 30. When the adjusting screw is turned, the lugs follow the contoured grooves thereby controlling the angle of the throttle plate. The valve shown in Figures 6 and 7 may also be modified to use a rectangular piston, similar to that shown in Figures 1 and 2. The piston may also take various other prismatic and non-prismatic shapes.

Instead of using a curved throttle plate, the piston may be provided with a curved throttle surface. In both cases, a tapered throttle passageway will provided and the valve will operate substantially as described above to provide a substantially constant flow rate.

Instead of adjusting the throttle plate in a direction perpendicular to the direction in which the piston moves, it may be arranged to be adjusted parallel to that direction. As the throttle passageway is tapered, this will increase or decrease the size of the throttle orifice, thereby increasing or decreasing the flowrate. Alternatively, the movement of the throttle plate may include both parallel and perpendicular components.

In yet another embodiment, the piston may be arranged centrally, with two or more adjustable throttle plates arranged radially around the piston.

Furthermore, the adjustable throttle plate may be replaced by a fixed throttle plate to provide a non-adjustable constant flow valve, which provides a pre-calibrated flow rate.

CLAIMS

1. A constant flow valve including a housing containing a fluid flow passageway, a resiliently-biased piston located in said passageway and mounted for movement in response to changes in the differential fluid pressure across the valve, a throttle member located in the passageway in the vicinity of the piston to define a throttle passageway between the piston and the throttle member, said throttle passageway being tapered and defining at its point of minimum size a throttle orifice, wherein the size of said throttle orifice varies in use with movement of the piston in response to changes in the differential pressure, such that the flow rate of fluid through the valve is substantially constant.

2. A constant flow valve according to claim 1, wherein the taper of the throttle passageway is non-linear.

3. A constant flow valve according to claim 1 or claim 2, including a flow adjusting means for adjusting the flow rate of fluid through the valve.

4. A constant flow valve according to claim 3, in which the flow adjusting means is connected to the throttle member and is operable to adjust the position of the throttle member relative to the piston.

5. A constant flow valve according to claim 4, in which the throttle member is adjustable in a direction substantially perpendicular to the direction of movement of the piston.

6. A constant flow valve according to any one of claims 3 to 5, in which the flow adjusting means is connected to the piston and is operable to adjust the position of the piston relative to the throttle member.

7. A constant flow valve according to claim 6, in which the piston is adjustable in a direction substantially parallel to the direction of movement of the piston.

8. A constant flow valve according to any one of the preceding claims, including profile adjusting means for adjusting the profile of the tapered throttle passageway.

9. A constant flow valve according to claim 8, in which the profile of the tapered throttle passageway is adjusted by adjusting the angle of the throttle member.

10. A constant flow valve according to claim 9 when dependent on claim 4 or claim 5, in which the profile adjusting means is arranged to adjust the angle of the throttle member as the position of the throttle member is adjusted.

11. A constant flow valve according to claim 8, in which the profile of the tapered throttle passageway is adjusted by adjusting the angle of the piston.

12. A constant flow valve according to claim 11 when dependent on claim 6 or claim 7, in which the profile adjusting means is arranged to adjust the angle of the piston as the position of the piston is adjusted.

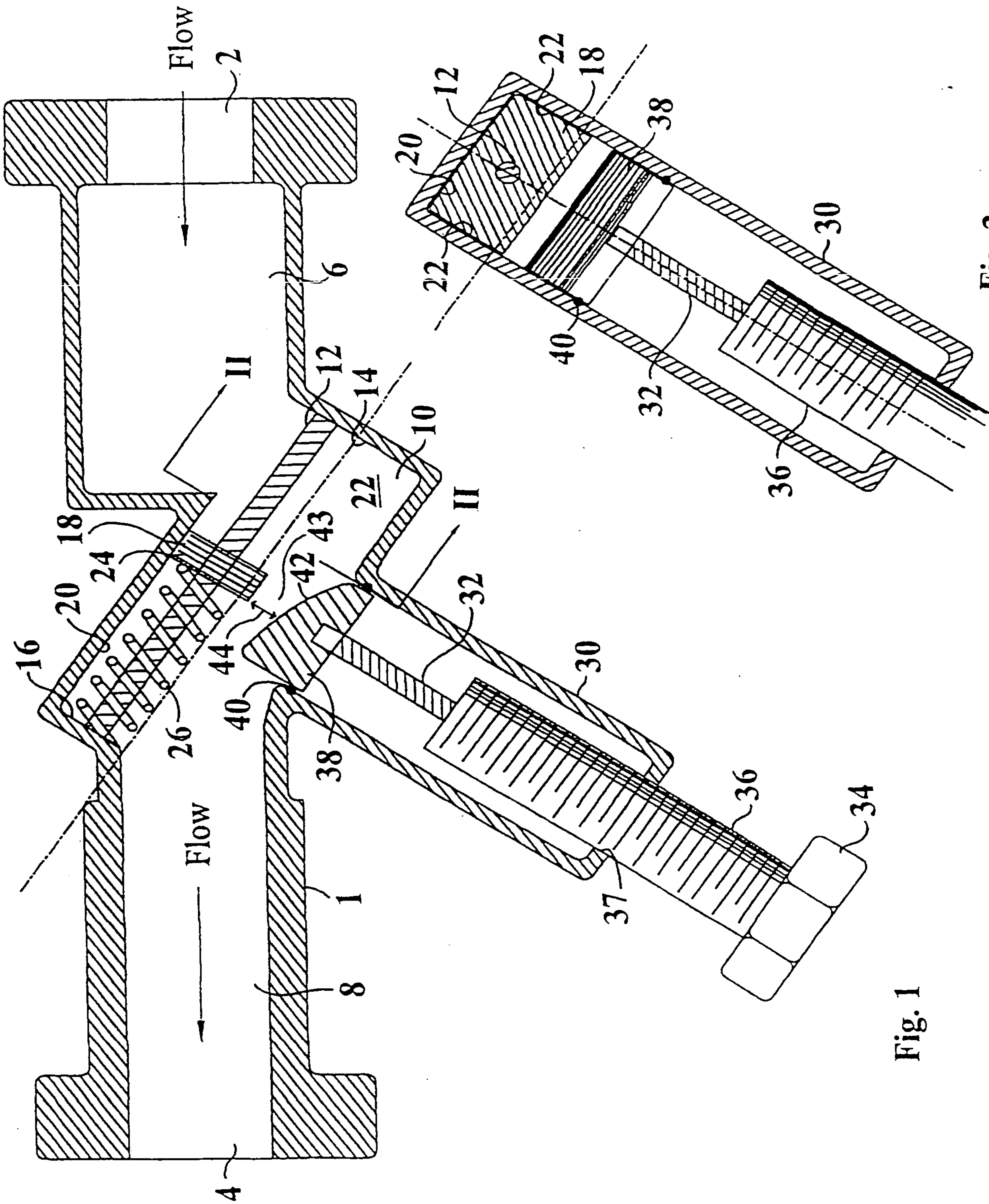


Fig. 1

Fig. 2

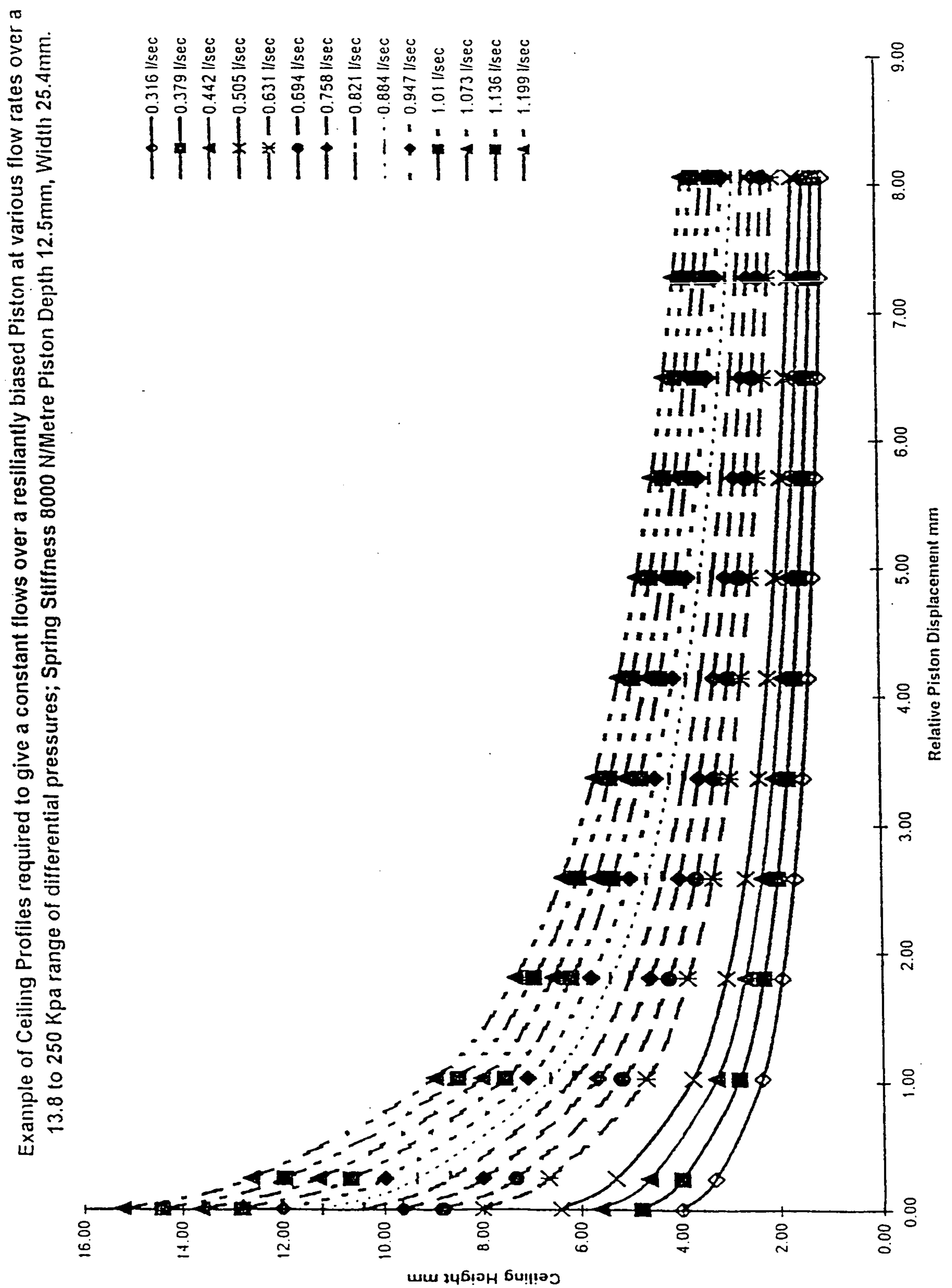

Example of Ceiling Profiles required to give a constant flows over a resiliantly biased Piston at various flow rates over a 13.8 to 250 Kpa range of differential pressures; Spring Stiffness 8000 N/Metre Piston Depth 12.5mm, Width 25.4mm.
0.316 l/sec
0.379 l/sec
0.442 l/sec
0.505 l/sec
0.631 l/sec
0.694 l/sec
0.758 l/sec
0.821 l/sec
0.884 l/sec
0.947 l/sec
1.01 l/sec
1.073 l/sec
1.136 l/sec
1.199 l/sec
Ceiling Height mm
16.00
14.00
12.00
10.00
8.00
6.00
4.00
2.00
0.00
Relative Piston Displacement mm
0.00
1.00
2.00
3.00
4.00
5.00
6.00
7.00
8.00
9.00

Fig. 3

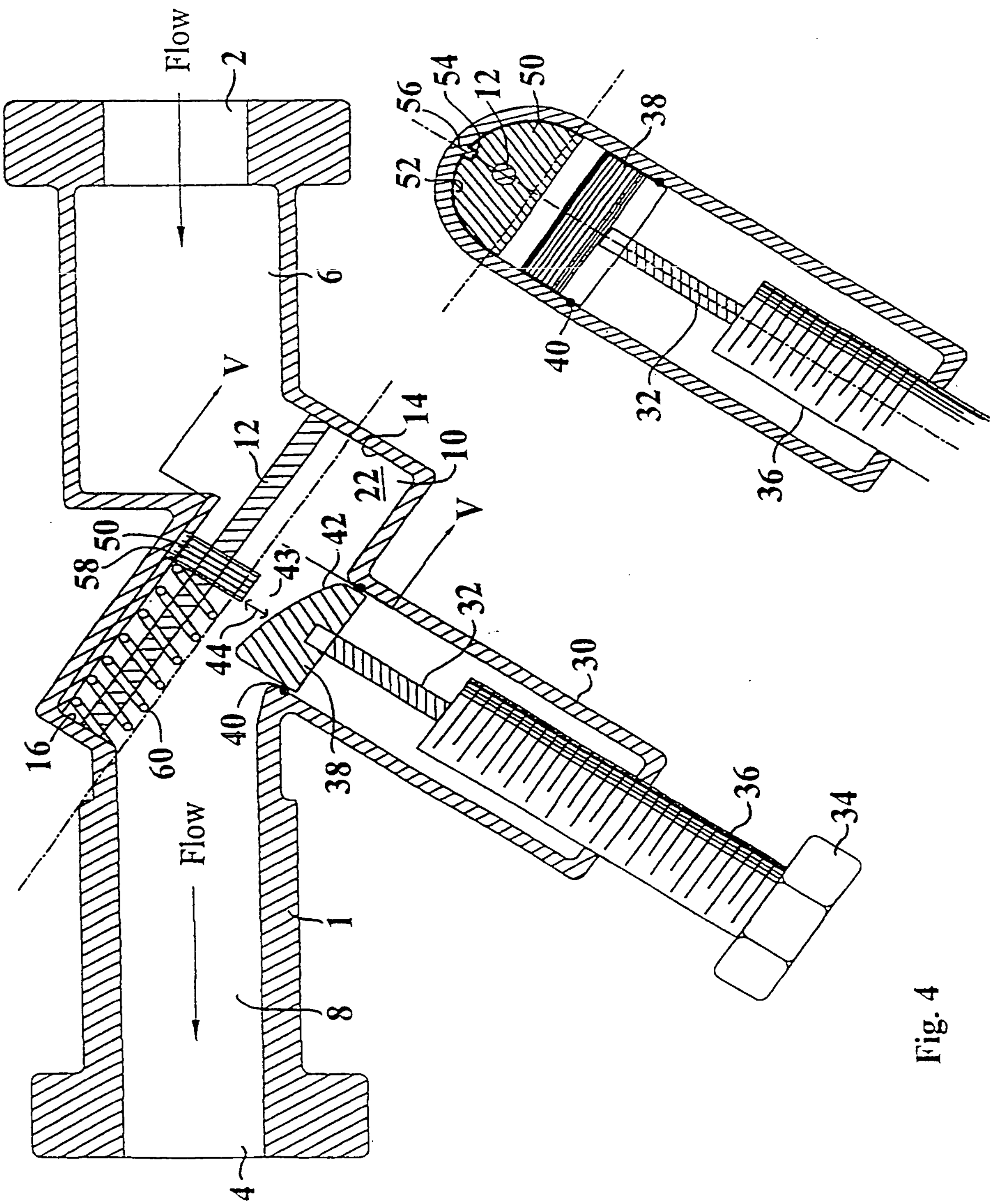

Flow
2
6
V
12
14
10
22
42
43'
50
58
44
16
60
40
38
32
30
36
34
1
8
Flow
4
V
56
54
12
50
52
38
40
32
36

Fig. 4

Fig. 5

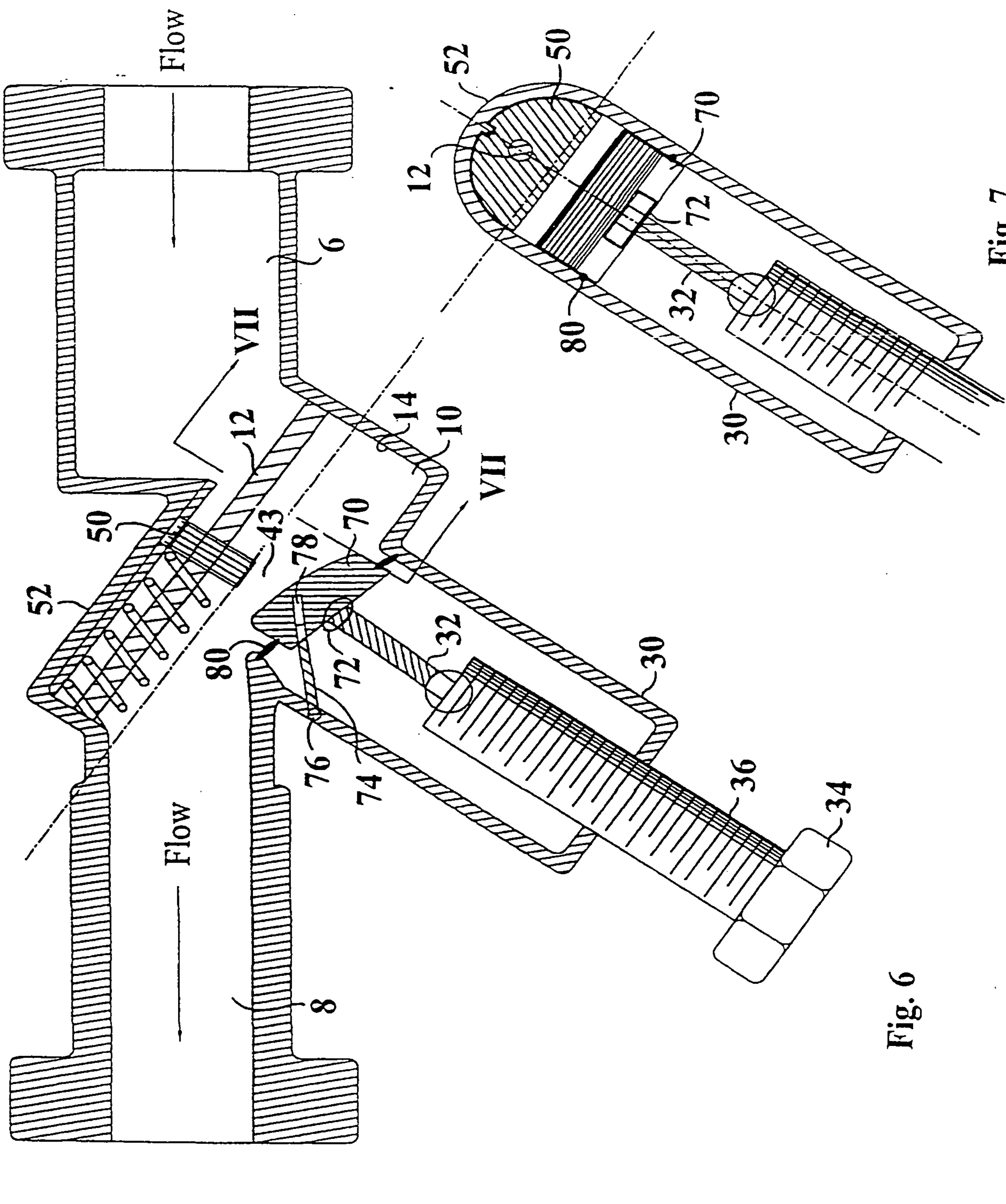

Flow
6
VII
12
14
10
VII
50
43
78
70
52
32
30
80
72
76
74
36
34
Flow
8
52
50
70
12
72
32
80
30

Fig. 7

Fig. 6

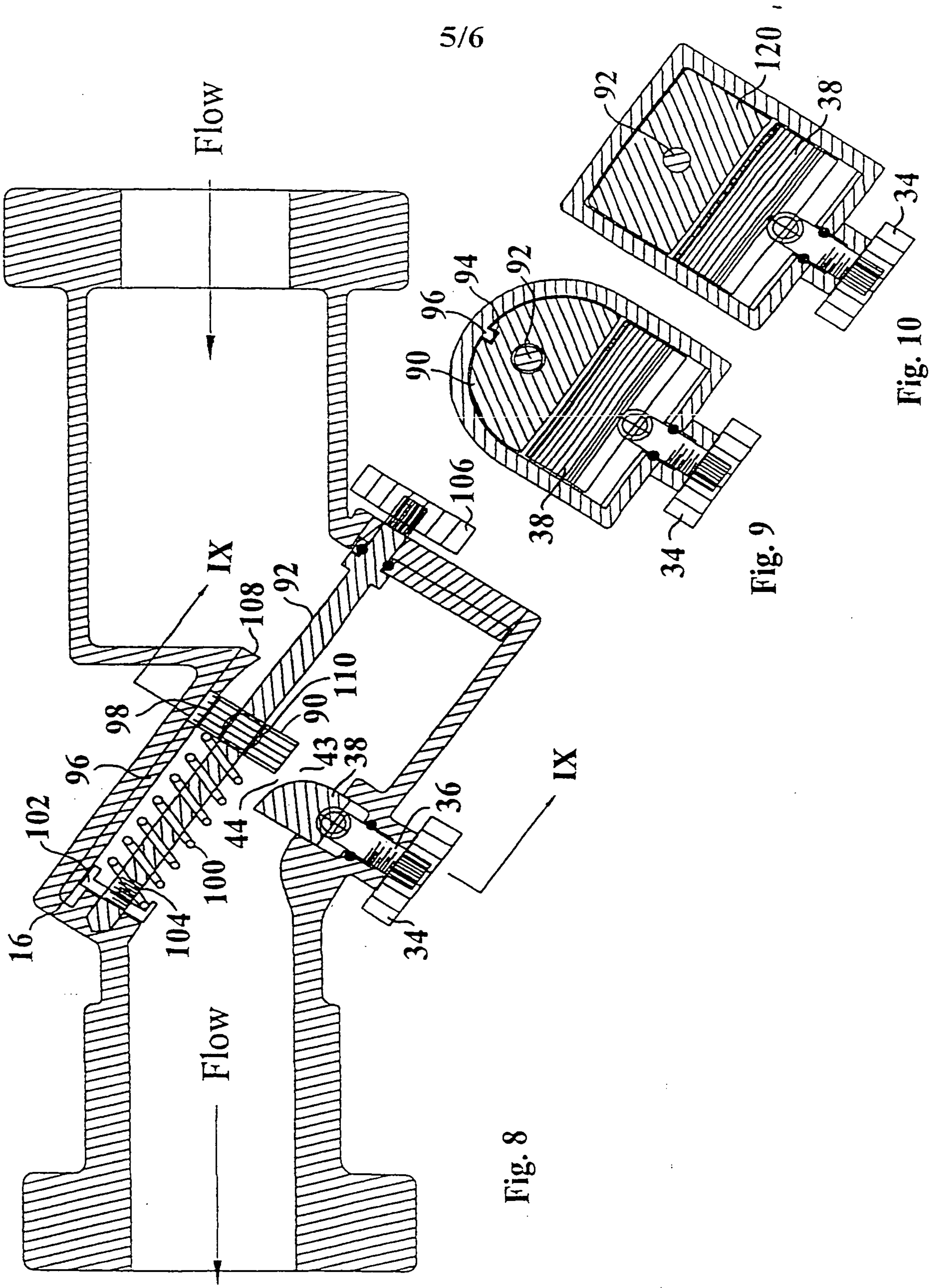


Fig. 8

Fig. 9

Fig. 10

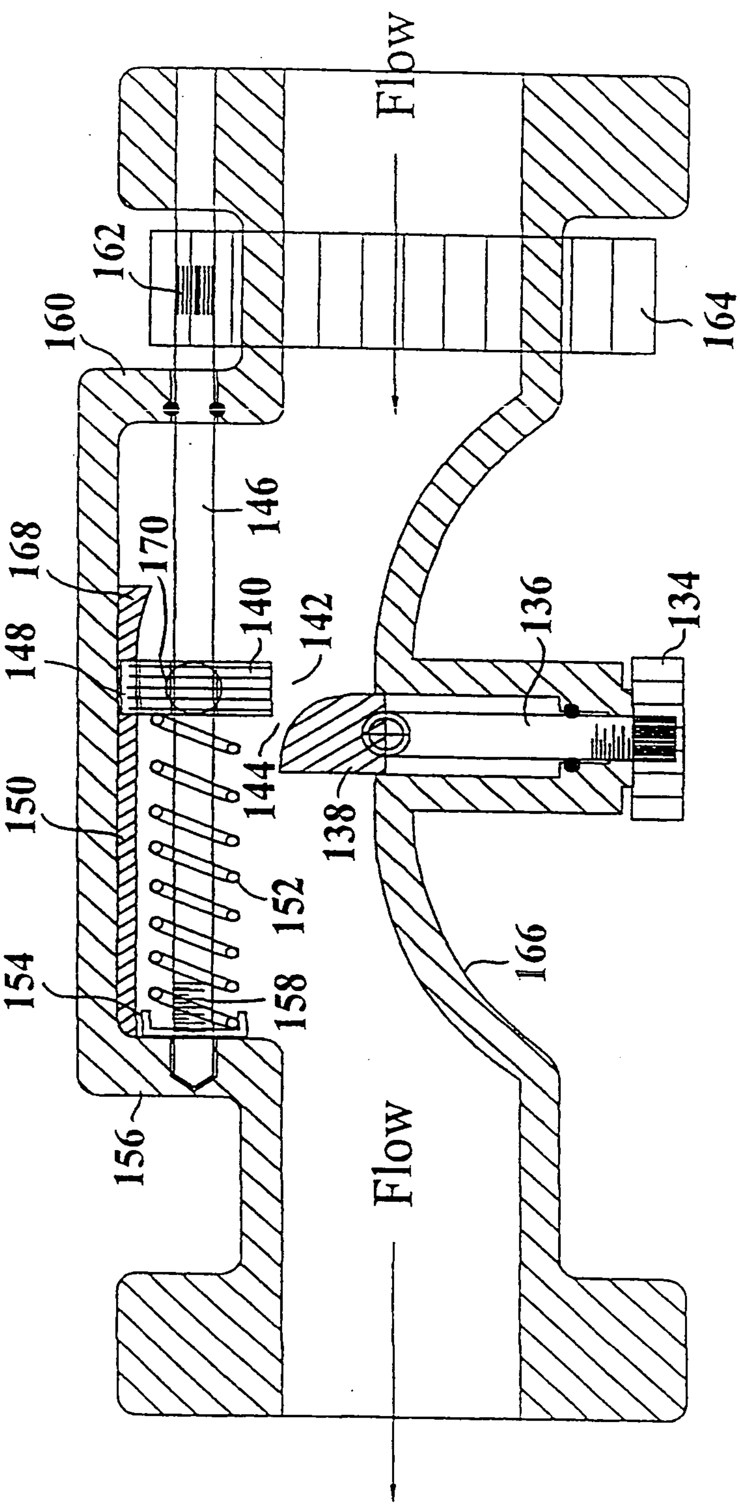

Flow
162
160
164
146
170
168
140
142
136
134
148
144
150
138
152
166
154
158
156
Flow

Fig. 11

INTERNATIONAL SEARCH REPORT

International Application No
PCT/GB 99/03597

A. CLASSIFICATION OF SUBJECT MATTER
IPC 7 G05D7/01

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 G05D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category *	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 3 464 439 A (T. BUDZICH) 2 September 1969 (1969-09-02) column 2, line 57 -column 4, line 62; figure 1	1-4,6
A	US 2 198 487 A (M.A. SISK) 23 April 1940 (1940-04-23) page 3, line 43 - line 61; figures 1-3	1
A	US 3 424 196 A (V.P. DONNER) 28 January 1969 (1969-01-28) column 3, line 55 -column 4, line 33; figures 1-3	1

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[X] Further documents are listed in the continuation of box C.

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26 January 2000	02/02/2000

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INTERNATIONAL SEARCH REPORT

Inte. onal Application No
PCT/GB 99/03597

C.(Continuation) DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication,where appropriate, of the relevant passages	Relevant to claim No.
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INTERNATIONAL SEARCH REPORT

Information on patent family members

Inte   onal Application No
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